WALLACE

Design of a Jib Crane

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DESIGN OF A JIB CRANE

BY

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

William Arthur Wallace

ENTITLED Design of a Jib Crane

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Bachelor of Science in Mechanical Engineering.

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THE DESIGN OF A JIB CRANE.

Chapter 1

Specifications

1. General Description- The type of crane designed is known as the single under braced jib and is to fulfill the following specifications:

Power - Electric

Capacity - Maximum hoisting load, six tons.

Dimensions - Height of mast from top of foundation to bottom of roof truss, twenty one feet.

Length of jib from center of mast to extreme end, eighteen feet.

2. Design- The frame of this crane is to be composed of structural shapes with gusset plate connections. The mast members to be bolted at top and bottom to heavy head castings; these castings to be fitted with forged steel pins properly centered and machined.

The top pin is to be connected with the top bearing casting which is to be designed for attachment to ceiling or truss of building.

The bottom pin is to turn in a heavy foundation bearing: the bottom of the pin resting on an anti-friction washer.

The jib end casting is to be fitted with a suitable idler sheave and attachment for securing the end of the hoist chain or wire rope.

Factor of Safety: This crane will be proportioned so that the factor of safety will be in no case less than five (5)

when the crane is operated at its full capacity. This factor is based on the ultimate strength of the material used.

3.- Material- The frame and miscellaneous structural material will be made of medium steel to conform to the specifications adopted by The Association of American Manufacturers. All iron castings will be of tough gray metal, free from injurious cold shuts or blow-holes.

4. Hoisting-Mechanism- The hoisting mechanism is to consist of a self-contained winch attached to, but practically independent of the crane frame. This winch is to be operated by a variable speed motor.

Shaft bearings throughout to be babbited and provided with suitable provisions for lubrication.

The drum is to be machine turned and grooved and of ample size to take without overlapping the effective length of the hoisting chain.

5. Brake- A despatch brake is to be supplied as a part of the hoisting mechanism designed for retaining the load at any position, or permitting it to be lowered rapidly under brake control at the will of the operator.

6. Racking Mechanism- The gearing for racking the trolley is to consist of two sprocket chain sheaves carrying a sprocket chain of uniform pitch, attached to the trolley and around on idler sheave at the end of the jib for pulling the trolley in either direction. This chain shall have provision for taking

up slack and be operated by a motor.

7. Trolley- The frame is to be supported by four flanged chilled wheels which travel on the upper flanges of the channels forming the jib. The trolley is to be fitted with extra large idler sheaves having turned grooves to insure smooth running of the hoisting chain. The trolley and block sheaves are to be bushed with bronze bushing.

8. Block- The block is to be of neat design having a heavy forged hook with head designed to turn on hardened steel ball bearings or bronze washer. The sheave of the block is to be of large diameter having turned grooves.

9. General- The crane throughout is to be designed in a high grade manner, all details being in harmony with the use of cut gearing etc., adapting the crane for service with high speed power at any time without replacing the gearing.

Chapter II

Frame Work.

10. Analysis of Forces- The forces acting in the various members of the crane were determined by the graphical method as follows:. Fig. 1 shows the crane indicated by the center lines of its members, and Fig. 2 the force polygon. The full lines indicate the forces on the members, when the trolley is at the maximum radius, and the dash lines when midway between the points M and N; letting AB represent the load, then BC and AD represent the horizontal pressures upon the top and bottom bearings respectively; B E the compression in the brace, and AF the force in the mast between the points N and L.

The results of the force polygon are:

Table 1

Member	Position of Load	
	Max Radius	Midway M&N
AB	12000	12000
BC AD	9440	3440
BE	21760	7840
AF	4560	5120

11. Jib Calculation- The maximum bending moment in the jib with the load midway between M and N is $12000 \times \frac{12}{4} = 36000$ ft. lbs. With the load at the maximum radius, the maximum bending moment is 12000 x 4.5 = 54000 ft. lbs. so this bending moment will be used in selecting a channel. Since the load is divided equally between two channels, each channel must be selected for a bending moment of 54000 ÷ 2 = 27000 ft. lbs. From a table of bending moments on page 109 in Cambria a C 41 - 12" - 30 lb. channel is selected which is rated to take a 28020 ft. lb. bending moment with a fibre stress of 12500 lbs. per square inch.

12. Stresses in the Jib- (a) Load midway between the points M and N.

Fig. 3 (a) shows the forces acting upon the jib, and Fig. 3 (b) the horizontal and vertical components of these forces. W represents the total weight of the jib, and W' that of the brace. Combining these known weights with the loads on the trolley and hoisting rope, R and T, the reactions of the mast and brace upon the jib may be determined. R_1 T_1 and R_2 T_2 are the horizontal and vertical components of these reactions.

Taking moments about the point P, we get

$$M = \frac{R_2 r}{2} - \frac{Wr^2}{8} = \frac{QZ}{n} + \frac{T_2 r}{2} - \frac{W'r}{4} - \frac{W(4L^2-3r^2)}{8} \quad (1)$$

The above equation is only approximate.

With the load Q at the point P the jib is in tension at the bottom and in compression at the top due to the moment M. Beside these stresses the jib has a compression uniformly distributed over the cross-section. This stress has a value.

$$\frac{\frac{Q}{n} - T_{,}}{2A} \qquad (2)$$

The resultant maximum compressive stress in the jib is then given by the following equation.

$$S = \frac{Mc}{2\,I} + \frac{\frac{Q}{n} - T_{,}}{2\,A} \qquad (3)$$

in which A represents the area and $\frac{I}{c}$ the section modulus of the cross-section of each member.

Substituting the numerical values of the different quantities in (1) and (3), we have the value of

M = 479040 in. lb.

S = 8787 pounds per square inch.

(b) Load at the maximum radius.

Fig. 3(c) shows all the forces acting upon the jib, and Fig. 3(d) the horizontal and vertical components of these forces. Proceeding as before R' and T', the reactions of the mast and brace upon the jib may be obtained; also their horizontal and vertical components. Taking moments about the point M, the bending moment in the jib is

$$M' = QM - \frac{QZ}{n} + \frac{WL^2}{2} \qquad (4)$$

The part of the jib to the right of the point M will be in compression at the bottom and in tension at the top, due to the moment M'. In addition to these stresses there is a compression uniformly distributed over the cross-section, given by the following expression:

$$\text{Compressive stress} = \frac{Q}{2nA} \qquad (5)$$

Between the points M and N the jib has a tensile stress equal to R', uniformly distributed over its cross-section.

The resultant maximum compressive stress in the jib is

$$S' = \frac{M'c}{2\,I} + \frac{Q}{2nA} \qquad (6)$$

Substituting the numerical values of the different terms in (4) and (6), the results are:

M' = 624960 in. lb.

S' = 11616 + 170 = 11786 lbs. per square inch.

13. Brace Calculations- The brace usually having considerable length must be considered as a long column and designed as such. Since it is rigidly attached at its ends, it would be classed as a "fixed ended column" or "square ended column". In computing the safe load that it will carry Gordon's formula is used. (See Cambria Hand Book). The brace being composed of two parts not laced or bound together at any point throughout their length, each member must be designed to carry half the total thrust. This thrust is a

maximum when the load is at the maximum radius, and is represented as T' in Fig. 3(c).

The brace is also subjected to a bending moment due to its own weight, but this is ordinarily small and is generally neglected.

Since the value of T' is 14080 pounds and is equally divided between the two channels, each channel must be designed for a load of $\frac{14080}{2}$ or 7040 pounds. Using a factor of safety of 5 each channel is treated as a "square ended column" having a load of 7040 x 5 = 35200 pounds.

Gordon's formula for "square ended columns" is

$$P = \frac{50000}{1+\frac{(12L)^2}{36000\ r^2}} \qquad (7)$$

in which P = ultimate strength in pounds per square inch, L = length in feet, and r = radius of gyration in inches.

From the table of properties of standard channels in Cambria Hand Book a C 25 - 8" - 11.25 lb. channel with a radius of gyration = 0.63 is chosen as having the proper section.

Substituting in Gordon's formula the values of the various terms

$$P = \frac{50000}{1+\frac{144 \times (18.416)^2}{36000 \times (1.63)^2}} = 11,337 \text{ lbs. per Sq. in.}$$

Since the cross-section of the channel is 3.35 square inches, the load the channel will carry is 3.35 x 11337 or 37980 lbs., so this section is selected, since the actual load is less than the allowable.

14. Mast Calculations- (a) Load at maximum Radius.

Neglecting the weights of the members, the external forces acting upon the mast are shown in Fig. 4(a), and their vertical and horizontal components in Fig. 4(b). For this loading the mast must be designed as a beam strong enough to resist the bending moment and thrust coming upon it. The maximum moment is on a section immediately above the line of action of R'_1 and its value is

$$M_1 = He - \frac{Q}{n} f \qquad (8)$$

Substituting numerical values in (8),

$M_1 = 35760$ ft, lbs.

Since the bending moment is equally divided between two channels, each must be designed for one half the bending moment or $\frac{35760}{2} = 17880$ ft. lbs.

Above the line of action R'_1 throughout the distance f the mast is in compression having a value

$$P = \frac{Q}{n} = \frac{12000}{4} = 3000 \qquad (9)$$

Using a fibre stress $S = 12500$ and $M = 17880$,

$\frac{I}{c} = \frac{17880}{12500} = 1.424$ the minimum section modulus that can be used. From the table of properties of channels, Cambria, a section C 41 - 12" - 20.50# is selected for symmetry and economy of material.

The maximum compressive stress will then be given by the following expression:

$$S_1 = \frac{M_1 c}{2\ I} + \frac{P}{2A} \qquad (10)$$

Substituting proper values in (10)

$$S_1 = \frac{35760}{2x1.75} + \frac{3000}{2x6.03} = 10466.4 \text{ lb./Sq. in.}$$

(b) Load near the mast.

With the load in this position the mast must be designed as a column to resist the thrust coming upon it. Fig. 4(c) represents the external forces coming upon the mast, and Fig. 4(d), the horizontal and vertical components of these forces neglecting the weight of the members.

Between the points K and N, the mast is subject to a compression equal to

$$P' = \frac{Q}{n} + R''_2 = \frac{12000}{4} + 6000 = 9000 \qquad (11)$$

Below K and down to the point L, the compression is R''_2

The resultant bending moment at the top is small for this loading andusually is not considered in the design.

A column designed for the load P'would not resist the bending moment when the load is at the maximum radius.

With the C 41 - 12" - 20.50# channel as chosen, the stress with the load near the mast is $\frac{9000}{2x6.03} = 745$ lbs./sq. in.

Chapter III

Block

15. Hook- The hook was designed from formulae given in Kent's Pocket Book, page 907 under the caption "Proportions of Hooks". A hook of 6 tons capacity was designed and the following dimensions as indicated by the figure 164 in the book calculated.

$A = 2\frac{1}{2}''$

$B = 2''$

$D = .5 \times 6 + 1.25 = 4.25 = 4\frac{1}{4}''$

$E = .64 \times 6 + 1.60 = 5.44 = 5\frac{7}{16}''$

$F = .33 \times 6 + .85 = 2.83 '' 2\frac{13}{16}''$

$G = .75 \times 4.25 = 3.18 = 3\frac{3}{16}''$

$H = 1.08 \times 2.5 = 2.70 = 2\frac{11}{16}''$

$I = 1.33 \times 2.5 = 3.32 = 3\frac{5}{16}''$

$J = 1.20 \times 2.5 = 3.00 = 3''$

$K = 1.13 \times 2.5 = 2.82 = 2\frac{13}{16}''$

$L. = 1.05 \times 2.5 = 2.62 = 2\frac{5}{8}''$

$M = .50 \times 2.5 = 1.25 = 1\frac{1}{4}''$

$N = .85 \times 2 - .16 = 1.54 = 1\frac{1}{2}''$

$O = .363 \times 6 + .66 = 2.84 = 2\frac{13}{16}''$

$Q = .64 \times 6 + 1.60 = 5.44 = 5\frac{7}{16}''$

$U = .866 \times 2.5 = 2.17 = 2\frac{3}{16}''$

16. Size of Hoisting Chain- In as much as the maximum load to be hoisted by the crane is 6 tons and the block is supported by 4 chains, each chain is assumed to be under a working load of

$$\frac{2000 \times 6}{4} = 3000 \text{ lbs.}$$

Newhall Chain Forge and Iron Co. B.B.B. Crane chain is selected and from a table given in Goodenough and Leutwiler's Mechanics of Machinery, P 37, a chain size $\frac{3}{8}$", pitch $\frac{31}{32}$", width of link $1\frac{1}{4}$", length of link $1\frac{3}{4}$", average weight per foot $1\frac{1}{2}$ lbs. proof test 4500 lbs., approximate breaking load 9000 lbs., and average safe working load 3000 lbs. is chosen.

17. Diameter of Sheaves- The diameter of sheaves in current practice is usually not less that 30 times the diameter of the chain. In order to keep the length of the trolley as small as possible, consistant with good practice, the pitch diameter of the sheave was made 12 inches which gives a ratio of sheave diameter to chain diamter of $12/\frac{3}{8} = 32$. The remainder of the sheave was proportioned in a substantial manner to produce symmetry.

18. Size of Block Pin- The block pin must be designed to resist bending and shearing for which a bending stress of 16000 lbs/sq. in. and a shearing stress of 10000 lb./sq. in. are assumed.

The pin with its loads comes under the classification of a beam supported at both ends with two symmetrical loads for which the maximum bending moment is $\frac{1}{2}$ Wa and the maximum shear $\frac{1}{2}$ W,

Where W= total load in lbs. = 12000

a= distance load to support = 1.81 in.

Then

Max. Bending Moment = $\frac{12000}{2}$ x 1.81 = 10860 in. lb.

$$\frac{I}{c} = \frac{M}{S} = \frac{10860}{16000} = 0.68 \qquad (12)$$

$$d = 1\frac{15''}{16}$$

Owing to the fact that cold rolled shafting comes in $\frac{1}{8}''$ sizes a 1 $\frac{7}{8}''$ pin will be used which gives under maximum load a stress of $\frac{10860}{0.659}$ = 16480 lbs.

The shearing value of this pin is area times shearing stress or 2.9483 x 10000 = 29483 lbs. while the maximum shear coming upon the pin is $\frac{1}{2}$ W= $\frac{12000}{2}$ = 6000 lbs.

19. Side Plates- The side plates are designed for a crushing stress of 20000 lb./sq. in. The thickness of the plate was determined from the bearing value which is equivalent to

$$D \times S \times t \qquad (13)$$

Where

D= diameter of pin in inches

S= crushing stress

t= thickness in inches

L= load in lbs.

Using a factor of safety of 5 and substituting numerical values

$$t = \frac{L \times 5}{D \times S} \qquad (14)$$

$$t = \frac{6000 \times 5}{1.875 \times 20000} = 0.8'' \text{ call } \frac{7''}{8}$$

Consequently to save material and reduce cost of crane the plates were made of $\frac{1''}{4}$ material reinforced with a strip $\frac{5''}{8}$ thick.

20. Swivel- The swivel is designed with a bronze ring. By considering the swivel, as a beam supported at both ends with a single concentrated load at the middle, see Fig. 5 the maximum bending moment is

$$M = \frac{P L}{4} \qquad (15)$$

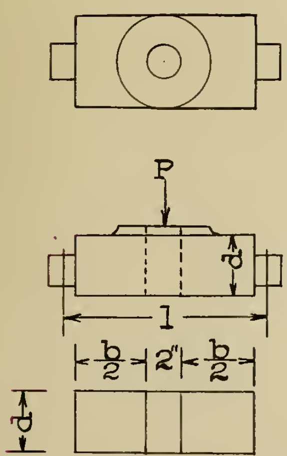


Fig. 5

Since P=12000 and L=8 $\frac{1''}{4}$

we find M=24750 in.lb.

From "Strength of materials" we get for the section in question,

$$\frac{M}{S} = \frac{bd^2}{12} \qquad (16)$$

Assuming b=5", we find

d=2.43"

21. Hoisting Chain Check- Assigning a value of 0.95 for the efficiency of a pulley including shaft friction, the efficiency of the mechanism of pulleys to where the chain winds onto the drum is, since five pulleys are used is $(0.95)^5$ or 0.774.

With the maximum load on the hook the force in the chain leading onto the drum is $\frac{3000}{.774} = 3870$ lbs: This is a little higher than the average safe working stress for a $\frac{3}{8}''$ chain but not enough to justify the use of a larger chain.

Chapter IV

Trolley

22. Trolley- The trolley was designed mainly by engineering judgment guided by builders photographs Ample strength is provided and the various component parts designed with the ease of manufacture and assembling in mind.

For the axles, the load was assumed to be similar to a beam supported at both ends with two symmetrical loads, so the maximum bending moment was

$$M = \frac{1}{2} Wa \qquad (17)$$

Where W = total load assumed 7000 lbs. per axle

a = distance from support to load, taken as 6.5 inches.

Assuming S = 16000 and substituting

$$\frac{M}{S} = \frac{I}{c} = \frac{.5 \times 7000 \times 6.5}{16000} = 1.42$$

This requires a diameter $d = 2\frac{7''}{16}$ but the axle was made $2\frac{1''}{2}$. The shearing value of this axle is Area x shearing Stress or 4.9087 x 10000 = 49087 lbs. and as the maximum shear is $\frac{1}{2}$ W, or $\frac{1}{2} \times 7000 = 3500$ lbs. the axle is of ample strength.

23. Racking Mechanism- The racking mechanism consists essentially of two chains, one on each side of the trolley, passing over idler sheaves and two corresponding

sprockets the latter actuated by a worm gear and motor.

The force required to move the trolley must be sufficient to overcome the following resistances:

(a) Journal friction of the trolley wheels.

(b) Rolling friction of the trolley wheels.

(c) Flange friction of the trolley wheels.

(d) Difference in load chain or rope tensions.

(e) Tension in the racking chain.

Notation:

Let Q = total load on trolley = 12000 lbs.

B = weight of block = 100 lbs.

W = weight of trolley = 700 lbs.

R = radius of trolley wheel

r = radius of trolley axle

Y = coefficient of journal friction (0.08 to 0.1)

c = coefficient of rolling friction (0.02 to 0.03")

A- Journal friction of the trolley axles.

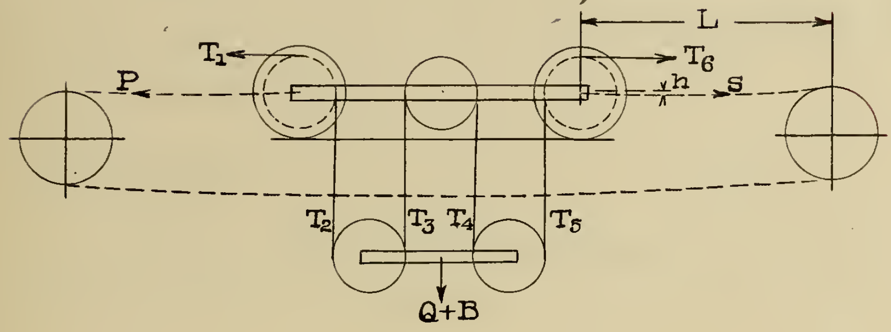


Fig. 6

Fig. 6 shows diagrammatically the arrangement of the hoisting and racking chain. From our theory of "Rope Stiffness", we have,

$$T_2 = k\ T_1$$
$$T_3 = k\ T_2 = k^2 T_1$$
$$T_4 = k\ t_3 = k^3 T_1$$
$$T_5 = k\ T_4 = k^4 T_1$$
$$T_6 = k\ T_5 = k^5 T_1$$

$$B+Q = T_2 + T_3 + T_4 + T_5 = kT_1\ (1+k+k^2+k^3)$$

$$\therefore \quad T_1 = \frac{(Q+B)(k-1)}{k(k^4-1)} \qquad (18)$$

$$\therefore \quad T_6 = \frac{k^4(Q+B)\ (k-1)}{(k^4-1)} \qquad (19)$$

The moment of friction upon the trolley axles

$$= ur\left[\sqrt{T_1^2 + \frac{(T_2 + T_3 + T_4 + W)^2}{2}} + \sqrt{T_6^2 + \frac{(T_5 + T_3 + T_4 + W)^2}{2}}\right] \qquad (20)$$

= ur z where z represents the bracket quantity. This moment must equal that due to the external force P_1 from which it follows that

$$P_1 = \frac{ur}{R} \quad (z) \qquad (21)$$

Substituting numerical values and assuming k = 1.05 we have

$$T_1 = \frac{(12000 + 100)\ (1.05 - 1)}{1.05\ (1.05^4 - 1)} = 2685 \text{ lbs.}$$

$$T_2 = 1.05 \times 2685 = 2820 \text{ lbs.}$$

$$T_3 = 1.05^2 \times 2685 = 2960 \text{ lbs.}$$

$$T_4 = 1.05^3 \times 2685 = 3110 \text{ lbs.}$$

$$T_5 = 1.05^4 \times 2685 = 3270 \text{ lbs.}$$

$$T_6 = 1.05^5 \times 2685 = 3430 \text{ lbs.}$$

$$z = 6761 + 7487 = 14,248$$

Substituting the value of z, p, R and r in (21)

P_1 = 356 lbs.

b- Rolling friction of the trolley wheels- From the theory of rolling friction we have for the force P_2 required to overcome this resistance the following:

$$P_2 = \frac{c}{R}\ (Q + W + B) \qquad (22)$$

Making the numerical substitutions

$$P_2 = \frac{.02 \times (12000 + 700 + 100)}{4} = 64 \text{ lbs.}$$

c- Flange friction of the trolley wheels-

According to Ernst's Hebezeuge, the flange friction may be taken as approximately 2% of the total load coming upon the trolley wheels. Calling this resistance P_3 we have

$$P_3 = 0.02\ (Q + W + B) \qquad (23)$$

Substituting we have

$$P_3 = 0.02\ (12000 + 700 + 100) = 256.\ \text{lbs.}$$

d- Difference in load chain or rope tensions.
The difference in tensions depends upon T_6 and T_1 and calling this difference P_4 we have

$$P_4 = T_6 - T_1 \qquad (24)$$

Substituting we have

$$P_4 = 3430 - 2685 = 745\ \text{lbs.}$$

e- Tension in the racking chain- This resistance is small and may be neglected ordinarily. If the racking chain is heavy the tension due to its own weight may be obtained by the use of the following formula:

$$S = \frac{W\,L^2}{8\,h} \qquad (25)$$

in which W represents the weight per foot of racking chain; L, the span in feet and h, the sag in feet.

f- The force required for racking the trolley:
The force P required for racking the trolley is equal to the sum of the various resistances given above, or

$$P = P_1 + P_2 + P_3 + P_4 + S \qquad (26)$$

Substituting the known values of the different P's

$P = 356.2 + 64 + 245 + 745 = 1210$ lbs.

24. Selection of racking chain- The type of chain decided upon for use is the Jeffrey-Mey-Obern link chain. Since there are two chains used on the trolley each chain will be uder a working stress of $\frac{1210}{2}$ or 605 lbs.
Consulting a Jeffrey chain catalogue we select a chain listed as Chactor #52 and having allowable working stress of 600 lbs.

Chapter V.

Winch.

25. Hoisting Drum- For use on the hoisting drum the "saw-buck" type of grooving was adapted. Using a pitch line diameter of 22" and an approximate length of chain to be wound up as 64 ft., the least number of grooves required, adding 3 extra turns, is

$$3 + (64/_{2\pi} \tfrac{11}{12}) = 3 + 11 = 14 \text{ grooves.}$$

For the saw-buck grooving, allowing $\frac{1}{8}$" clearance between wraps, a $\frac{3}{8}$" chain requires the grooves to be $1\frac{3}{16}$" center to center, which makes the least length of drum required as

$$14 \times 1\tfrac{3}{16}" = 16\tfrac{5}{8}" \text{ long.}$$

The proportions of the drum were made in accordance with current practice as adapted by the leading crane builders.

26. Design of gearing- The gearing ratios and diagrammatic arrangement are shown in the accompanying diagram.

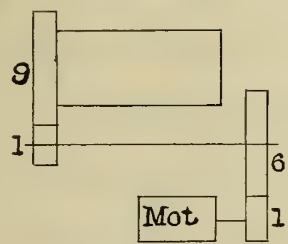


Fig. 7

Since the force on the chain leading onto the drum is 3430 lbs. and the pitch line diameter of the drum is 22", the torque required on the motor pinion is

$$3430 \times \frac{11}{12} \times \frac{1}{9} \times \frac{1}{5} = 69.8 \text{ ft. lbs.}$$

Taking a motor pinion of 6" diameter the value of the force W exerted on the tooth face is

$$69.8 \times \frac{12}{3} = 279.2 \text{ lbs. call it 280 lbs.}$$

Using cut teeth and a diametral pitch of 5, the number of teeth is 30, the circular pitch 0.6283 and the value of Y from chart in "Machine Design Notes" is 0.102.

Assuming the maximum speed to be 1178.'/min. the value of the stress S as obtained from chart is 2700 lbs. Taking the face of the pinion as $1\frac{3}{4}$" and substituting in Lewis formula

$$W = S\ p'\ f\ y \qquad (27)$$

we have

$$W = 2700 \times 0.623 \times 1.75 \times 0.102 = 300 \text{ lbs.}$$

which will be satisfactory.

The intermediate pinion being 5" pitch diameter, the pitch line speed will be

$$1178 \times \frac{5}{30} = 194'/\text{ min.}$$

Assuming a diametrical pitch of 3 and a 4" face, the circular pitch is 1.0472, the number of teeth 15 and y, the strength factor, 0.076 from the chart. The stress as obtained from the chart is S = 6000 lbs.

The value of the force W exerted on the tooth face required $= 280 \times \frac{30}{5} = 1680$ lbs.

Substituting in formula we have

$$W = 6000 \times 1.0472 \times 4 \times 0.076 = 1670 \text{ lbs.}$$

The gears to mesh with the pinions are always stronger than the pinion and obviates the necessity of calculating them.

To sum up the gears the following table, Table II is added.

Table II

Gear	Dia.	P	f	Mat.
Motor pinion	6	5	$1\frac{3}{4}$	C 1
Intermediate	30	5	$1\frac{3}{4}$	C 1
Intermediate pinion	5	3	4	C 1
Drum Gear	45	3	4	C 1

27. Proportions of gears- The proportions of the intermediate gear are as follows.

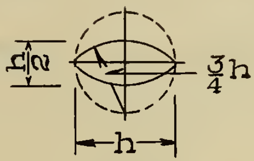


Fig. 8

Thickness of rim $= \frac{p'}{2} =$

$\frac{0.6283}{2} = .314"$ call $\frac{3}{8}"$

Thickness of reinforcing rib $= \frac{2}{3}\ p' =$

$\frac{2 \times 0.6283}{3} = .418"$ call $\frac{1}{2}"$

$$h = \sqrt[3]{\frac{20\ T}{N\ S}} \qquad (28)$$

Where N = number of arms = 6

T = torque = 280 x 15

S = allowable stress = 1000

Substituting in the above formula we have

$$h = \sqrt[3]{\frac{20 \times 280 \times 15}{6 \times 1000}} = 2.4" \text{ call it } 2\frac{1}{2}"$$

$$\frac{h}{2} = \frac{2\frac{1}{2}}{2} = 1\frac{1}{4}"$$

The drum gear dimensions are calculated in a similar manner and the following results are obtained.

n = 6

S = 2000

then

$$h = 4''$$

$$\frac{h}{2} = 2''$$

$$\text{thickness of rim} = \frac{1''}{2}$$

$$\text{thickness of reinforcing rib} = \frac{3''}{4}$$

28. Shafting- Since in raising and lowering the load, the position of the chain varies from one end of the drum to the other, the maximum bending moment will be when the chain is at the center of the drum. On this basis the

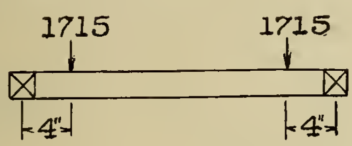


Fig. 9

drum shaft will be classified as a beam supported at both ends with two symmetrical loads as shown in Fig. 9. For this style of loading the maximum bending moment

$$M = \frac{1}{2} Wa \qquad (29)$$

Where

W = total load in lbs. = 3430

a = distance from load to support = 4"

Substituting we have

$$\text{Max. B.M} = \frac{3430 \times 4}{2} = 6860 \text{ in. lbs.}$$

Neglecting friction the shaft also has a twisting moment of

$$3430 \times 11 = 37730 \text{ in. lbs.}$$

By Guest's Law, $M_e = \sqrt{T^2 + M^2}$ (30)

Where

T = twisting moment

M = bending moment

We have

$$M_e = \sqrt{(37730)^2 + (6860)^2} = 38348 \text{ in. lbs.}$$

Assuming $S = 10000$ we have

$$\frac{I}{c} = \frac{Me}{S} = \frac{38348}{10000} = 3.84 \qquad (31)$$

from which $d = 3\frac{3}{8}''$

The intermediate shaft is subjected to a twisting moment

$$1670 \times 2.5 = 4175 \text{ in. lbs.}$$

The maximum bending moment is W L due to loading being that of a cantilever beam where

W = total load in lbs. = 1670

L = distance from load to support = 4"

Substituting these values we have

Max B.M = 1670 x 4 = 6680 in lbs.

Using Guests Law, M_e = 7880 in Lb.

Assuming $S = 10000$, we find $D = 2''$

29. Despatch Brake- The absorbed energy is made up of the following parts: (1) The work done by the live load; (2) the kinetic energy of the rotating parts.

The work done by the live load in a unit of time is Q V and its kinetic energy $E_1 = \frac{QV^2}{2\,Q}$; therefore in coming to rest

in t seconds the live load requires an expenditure of

$$W + E_1 \text{ ft. lb. of work} \qquad (32)$$

Where

$$W = Q\,V\,T \qquad (33)$$

Q = live load on hoisting drum

V = linear velocity of drum in ft. per sec.

t = number of seconds brake is applied.

Substituting known values in above equation we get

$$E_1 = \frac{3420 \times 1.63 \times 1.63}{2 \times 32.2} = 141 \text{ ft. lbs.}$$

Allowing the number of seconds the brake is applied to be 5

$$W = 3420 \times 1.63 \times 5 = 27800 \text{ ft. lbs.}$$

$$E_1 + W = 141 + 27800 = 27941 \text{ ft. lbs.}$$

Allowing 5% for inertia of moving parts we have for the energy to be absorbed.

$1.05\,(E_1 + W) = 1.05 \times 27941 = 29338$ ft. lbs. call it 30000 ft. lbs.

Using the rule experience has shown that 1 sq. in. of friction surface for each 200 or 250 foot pounds of energy absorbed, we require in this case 30000/200 or 150 square inches of friction surface.

To provide this area of friction surface three fibre and two steel discs are used having rubbing surfaces of discs equal to the area contained between circles of four and eight inch diameters, so the total friction surface supplied =

$$5\,(50.26 - 12.56) = 188 \text{ sq. in.}$$

Frames- The frames are designed with a view of easy manufacture and with bearings placed at a 45° angle. The pressures on the bearing surface in pounds per square inch of bearing are within the limits of common usage.

Chapter VI

Miscellaneous

31. Pintles- The pintles are designed for a bearing pressure of 600 lbs. per sq. in. of projected area and must also be strong enough to resist the cross bending and shear that come upon them.

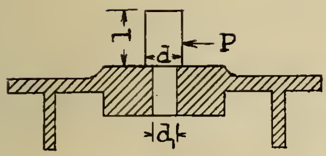


Fig. 10

The projected area $d \times L = \frac{9440}{600} = 15.8$ sq. in.

Assuming a pin $3\frac{1}{2}'' \times 4\frac{1}{2}$

$$d_1 = d - \frac{1}{2} = 3\frac{1}{2} = 3''$$

Treating the pin as a cantilever beam, Fig. 10, we find the stress

$$S = \frac{Mc}{I} = \frac{21300}{2.7} = 7800 \text{ lb. per sq. in.}$$

The bottom pintle must also resist the vertical load coming upon it, hence the area of the collar should be made large enough so as to keep the crushing stress of the casting at a rather low limit. For the proportions assumed above, this crushing stress is found to be 2020 lb. per sq. in. which is safe.

32. Riveting- For the size of main frame members used in the design, practice in structural iron work is to make all field rivets $\frac{3}{4}''$, hence that size will be used. The shearing value of this rivet in single shear is 2651 lbs.

For the jib and mast, we use force $R_1' = 14080$ which gives the minimum number of rivets required as 3.

Likewise for the jib and brace and mast and brace we use the forces $T_1' = 14080$ and $T_2' = 16560$ which gives 3 and 4 rivets required respectively.

More rivets than called for by the above will be used to fill out the joint and also to allow for 50% increase in number of rivets on account of field riveting.

33. Design of Worm Gear- Using a bronze gear of 4.78" pitch diameter and a circular pitch of $\frac{3}{4}$" we have since we assume the speed to be 20 ft. per min.

$S = 11600$ lbs.

$f = 2"$

$N = 20$ teeth

$y = .088$

$p' = 0.75$

$W = 1265$ lbs. required

Applying the Lewis formula

$W = 11600 \times 0.75 \times 2 \times .088 = 1530$ lbs.

The worm will be $2\frac{1}{2}$ inch diameter outside and made integral with the shaft, which is $\frac{15}{16}$" in diameter.

34. Motors- The motors selected are those of the Northern Electrical Manufacturing Co., Madison, Wis. designed specially for use on cranes.

Since the motor torque required on the hoist is 69.6 ft. lb., by consulting the characteristic curves of the motors, a 10 H P -"S" frame motor is selected.

For the racking mechanism the horse power required from the motor, basing the efficiency of the worm and gear at 45%, is 1.63 so a 1 $\frac{1}{2}$ H P - "P" frame motor is selected.

Bill of Material.

Cast Iron

No.	Name of Piece	No. Wanted	Plate No.
1	Pintle Casting	2	VIII
2	Pintle Bearing	2	VIII
3	Side Frame	2	VI
4	Drum	1	VI
5	Brake Casing	1	VII
6	Hand Wheel	1	VII
7	Disc Flange	1	VII
8	Trolley Wheel	4	V
9	Trolley Axle Bearing	2	V
10	Oil Pocket	4	V
11	Sheave	6	IV
12	Ratchet Wheel	1	IX
13	Bearing Cap	1	IX
14	Bearing Cap	1	IX
15	Motor Pinion	1	IX
16	Intermediate Gear	1	X
17	Drum Gear	1	X
18	Gear Case	1	X
19	Gear Case Cover	1	X

Steel.

No.	Name of Piece	No. Wanted	Plate
01	Jib & Brace Gusset Plate	2	III
02	Jib & Mast Gusset Plate	2	III
03	Brace & Mast Gusset Plate	2	III
04	Cross Bar	3	III
05	Worm	1	X
06	Drum Shaft	1	IX
07	Sprocket Shaft	1	IX
08	Intermediate Shaft	1	IX
09	Take-up Arm Rod	1	IX
010	Pawl	1	IX
011	Space bolt	4	IV
012	Pin lock	6	IV
013	Hook Trunnion	1	IV
014	Hook	1	IV
015	Side Plate	2	IV
016	Detail of Pin	1	IV
017	Trolley Axle	2	V
018	Sheave Pin	1	V
019	Side Plates	4	V
020	Oil Pocket Cover	4	V
021	Steel Disc	2	VII
022	Tie Bar	1	VI
023	Motor Brace	1	VI
024	Motor Brace	1	VI
025	Shaft Collar	1	VI
026	Stop Block	4	VIII

Steel (cont)

No.	Name of Piece	No. Wanted	Plate
027	Idler Pin	1	VIII
028	End Plate	1	VIII
029	Take-up Arm	4	VIII
030	Eye bolt	2	VIII
031	Idler Collar	2	VIII
032	End Plate Fastening	2	VIII
033	Brace	2	VIII
034	Brace	1	VIII
035	Pinion Collar	1	VIII
036	Pintle Pin	2	VIII

Bronze.

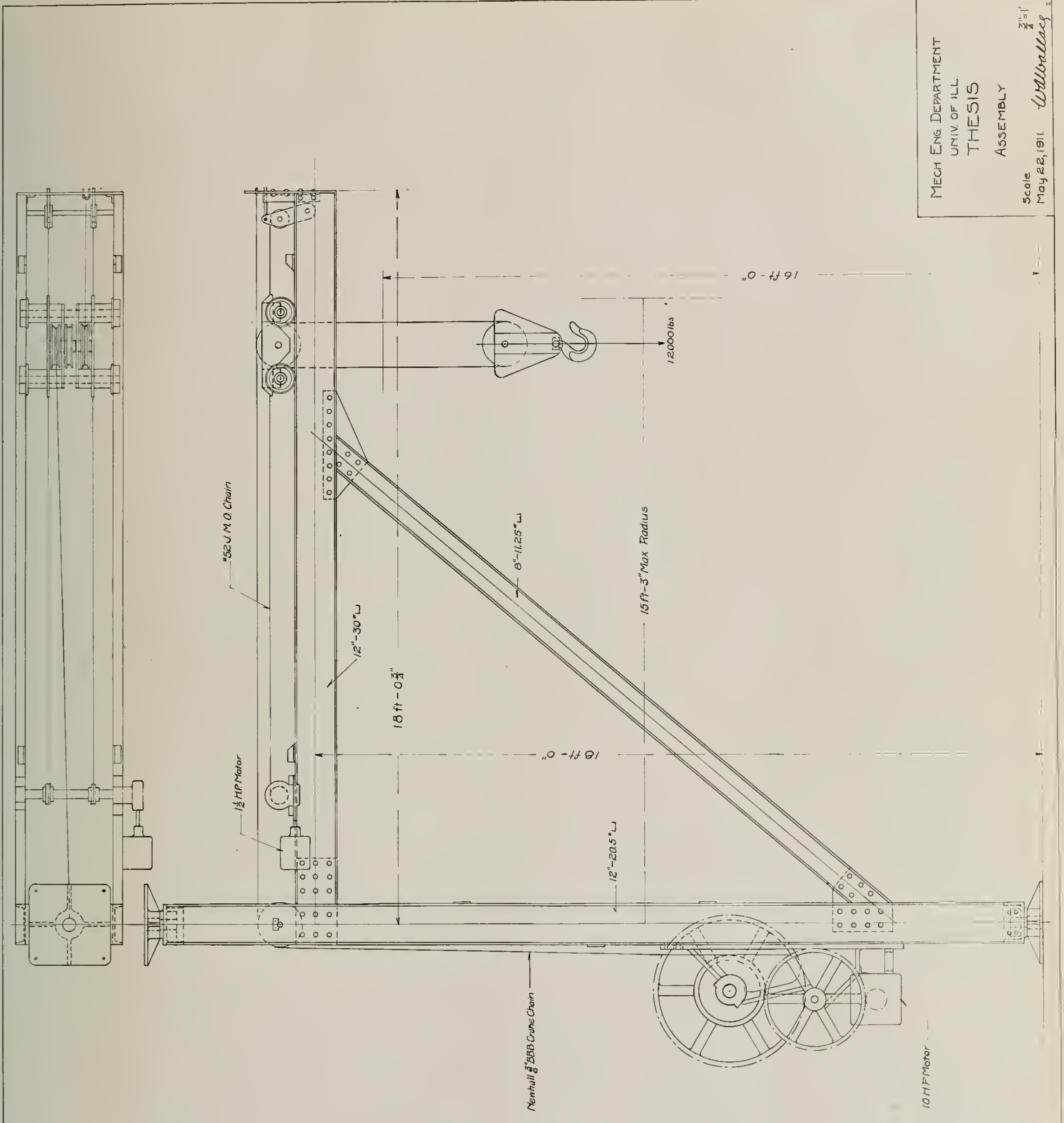
No.	Name of Piece	No. Wanted	Plate
001	Pintle Bushing	1	VIII
022	Sheave Bushing	6	IX
003	Brake Bushing	1	IX
004	Trolley Axle Bushing	4	IX
005	Swivel Ring	1	IX
006	Worm Gear	1	X
007	Thrust Collar	2	X

Wrought Iron.

No.	Name of Piece	No. Wanted	Plate
W 1	Take-up Collar	2	IX
W 2	Chain Anchor	1	VI

Fibre

No.	Name of Piece	No. Wanted	Plate
F 1	Fibre Disc.	3	VII


MECH ENG DEPARTMENT
UNIV. OF ILL.
THESIS
ASSEMBLY
Scale 3/4"=1'
May 22, 1911.
16 ft - 0"
12000 lbs
15 ft-3" Max Radius
8"-11.25"
#52 J M O Chain
12"-30"
18 ft - 0 3/4"
18 ft - 0"
1 1/2 HP Motor
12"-20.5"
Newhall 3/8" BBB Crane Chain
10 HP Motor
22 ft - 0"

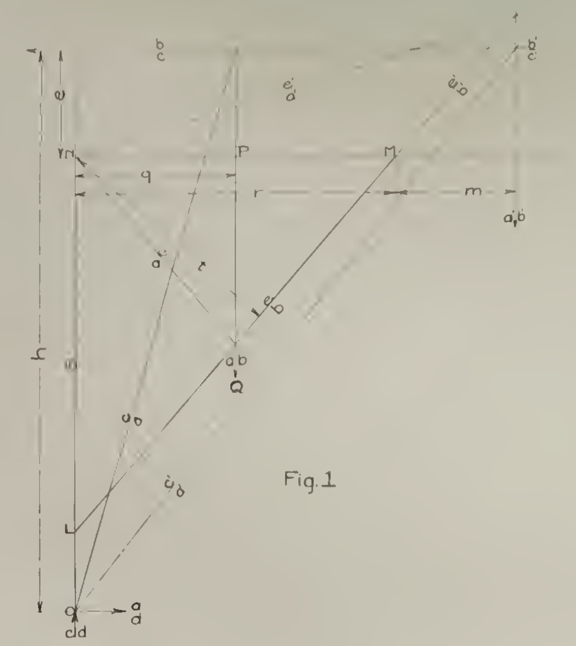


Fig. 1

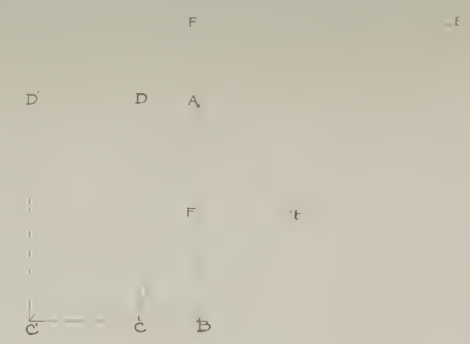


Fig. 2

Fig. 3

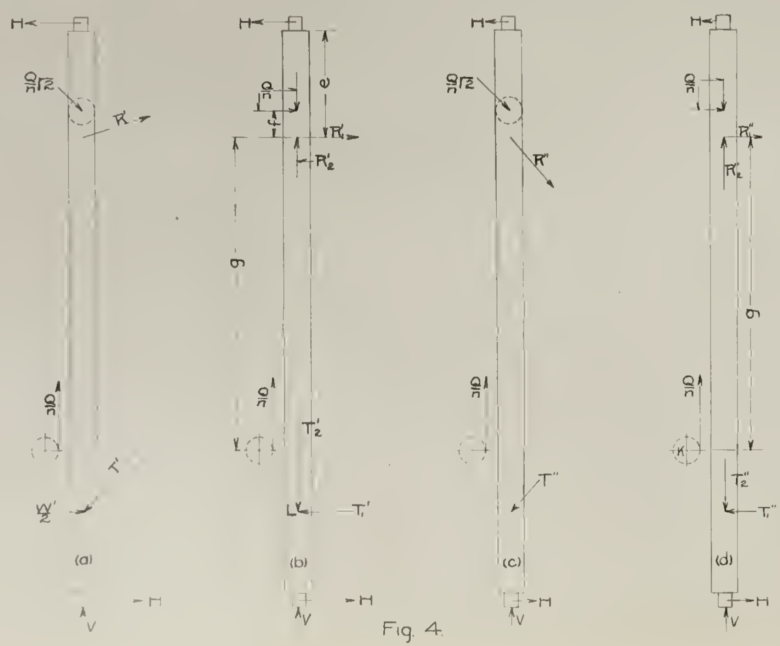
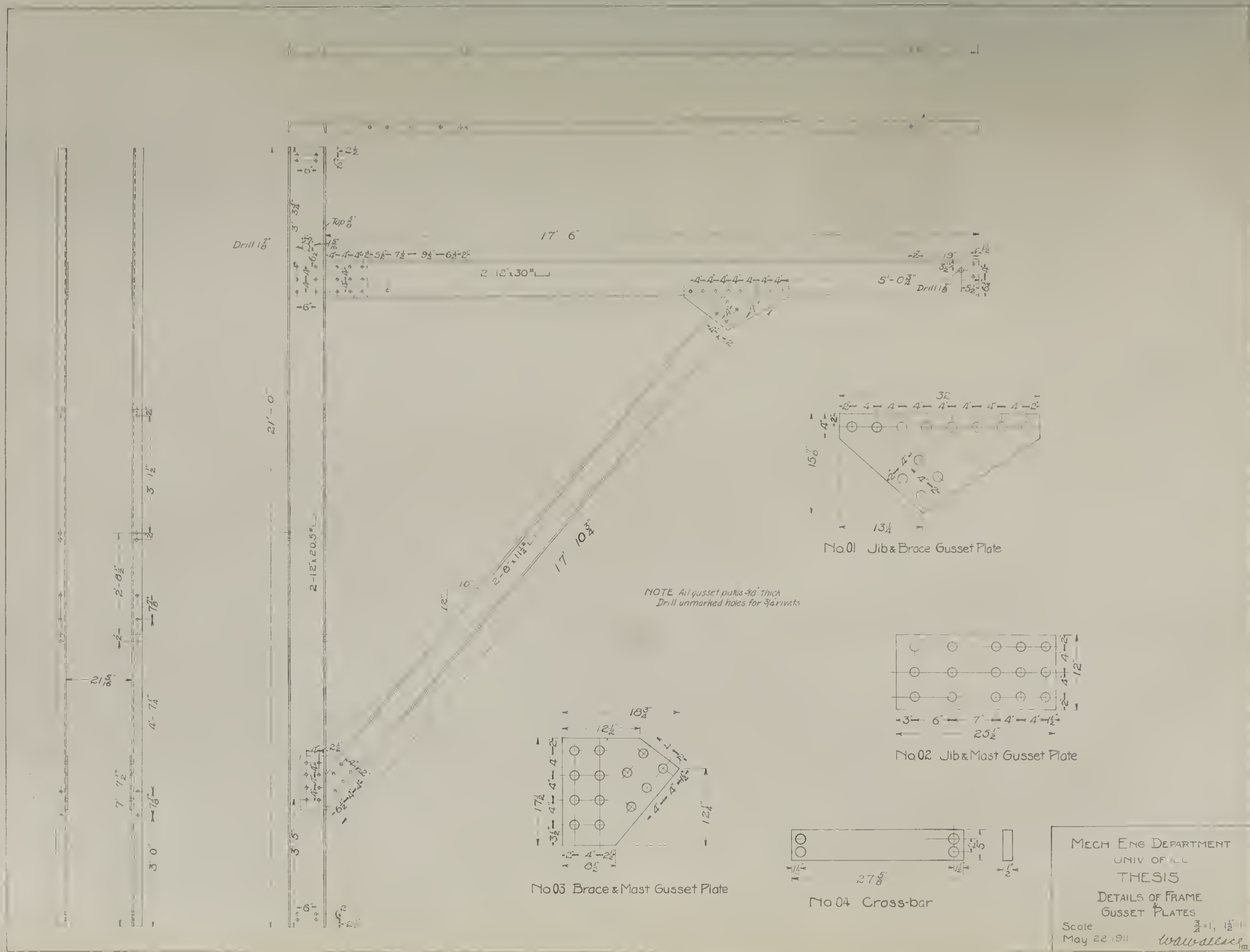
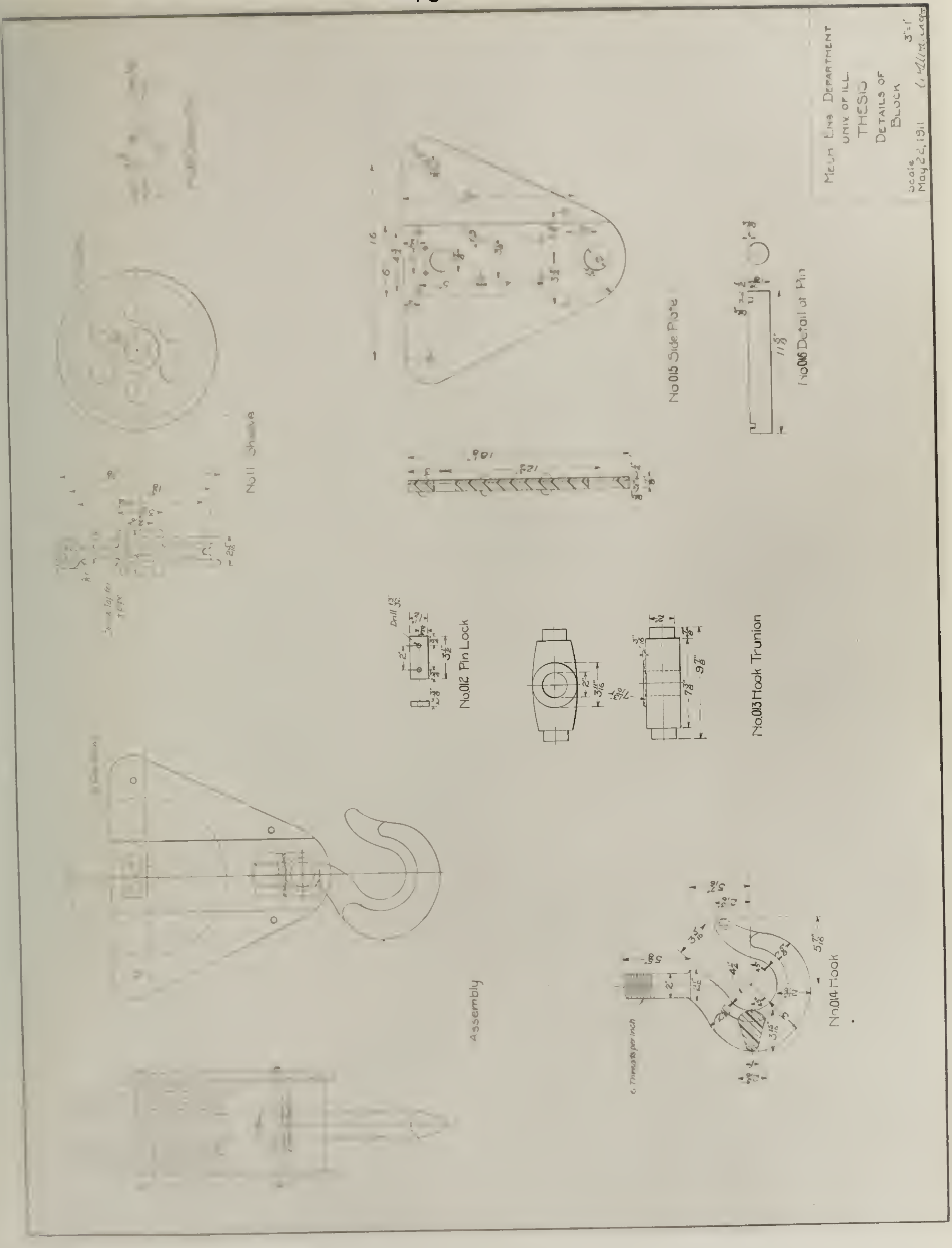
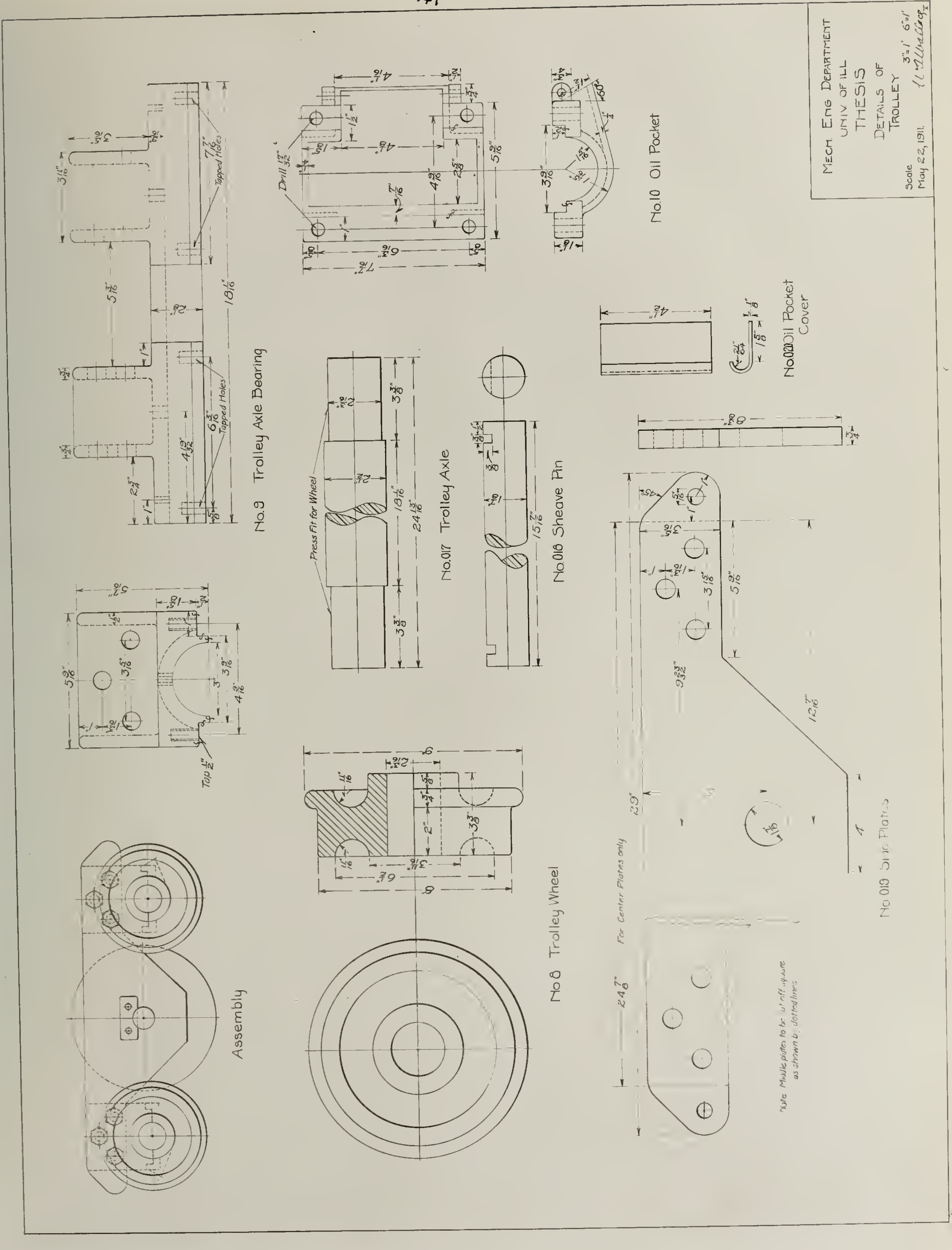


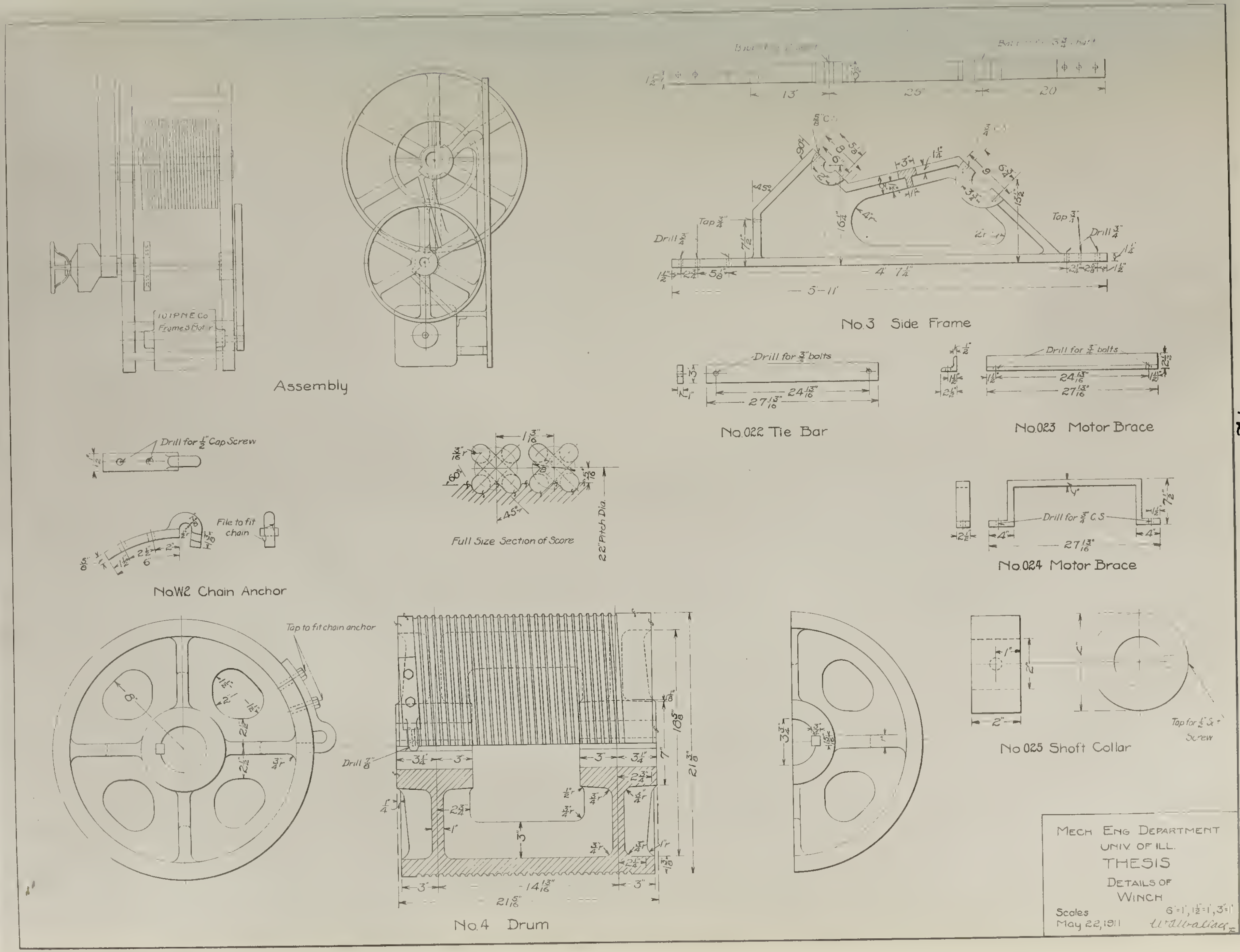
Fig. 4.

MECH. ENG DEPARTMENT
UNIV. OF ILL
THESIS
DIAGRAMS
&
POLYGON
Scales $\frac{3}{8}"=1'$ $1"=4000$ lb
May 22, 1911

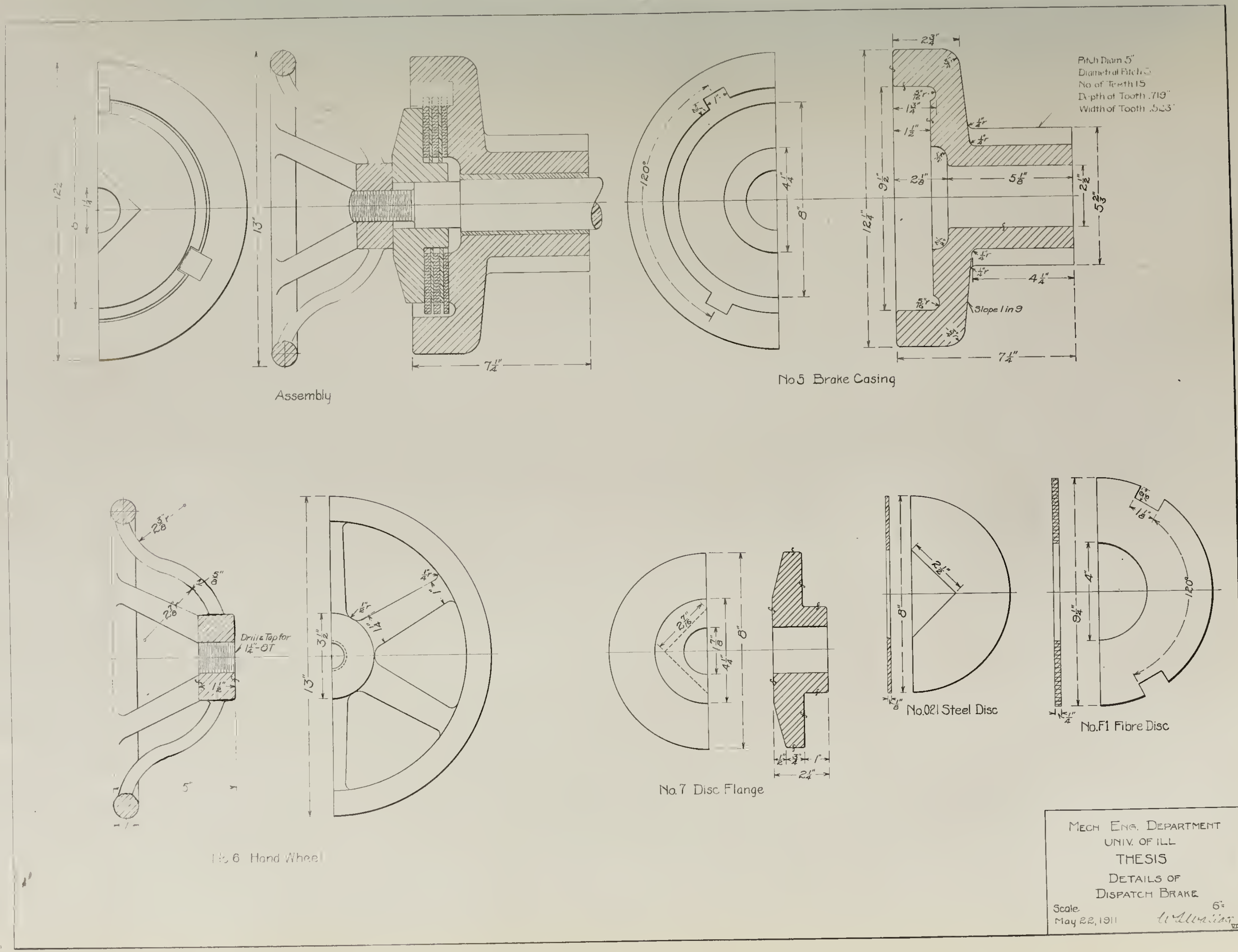

Mech Eng Department
Univ of Ill
Thesis
Details of Frame & Gusset Plates
Scale 3/4=1, 1 1/2=1
May 22 1911
No.01 Jib & Brace Gusset Plate
No.02 Jib & Mast Gusset Plate
No.03 Brace & Mast Gusset Plate
No.04 Cross-bar
NOTE All gusset plates 3/8" thick
Drill unmarked holes for 3/4 rivets
17' 6"
17' 10 3/4"
21'-0"
5'-0 3/4"
2 12"x30# [
2-12"x20.5# [
2-8"x11 1/4# [
Drill 1 1/8"

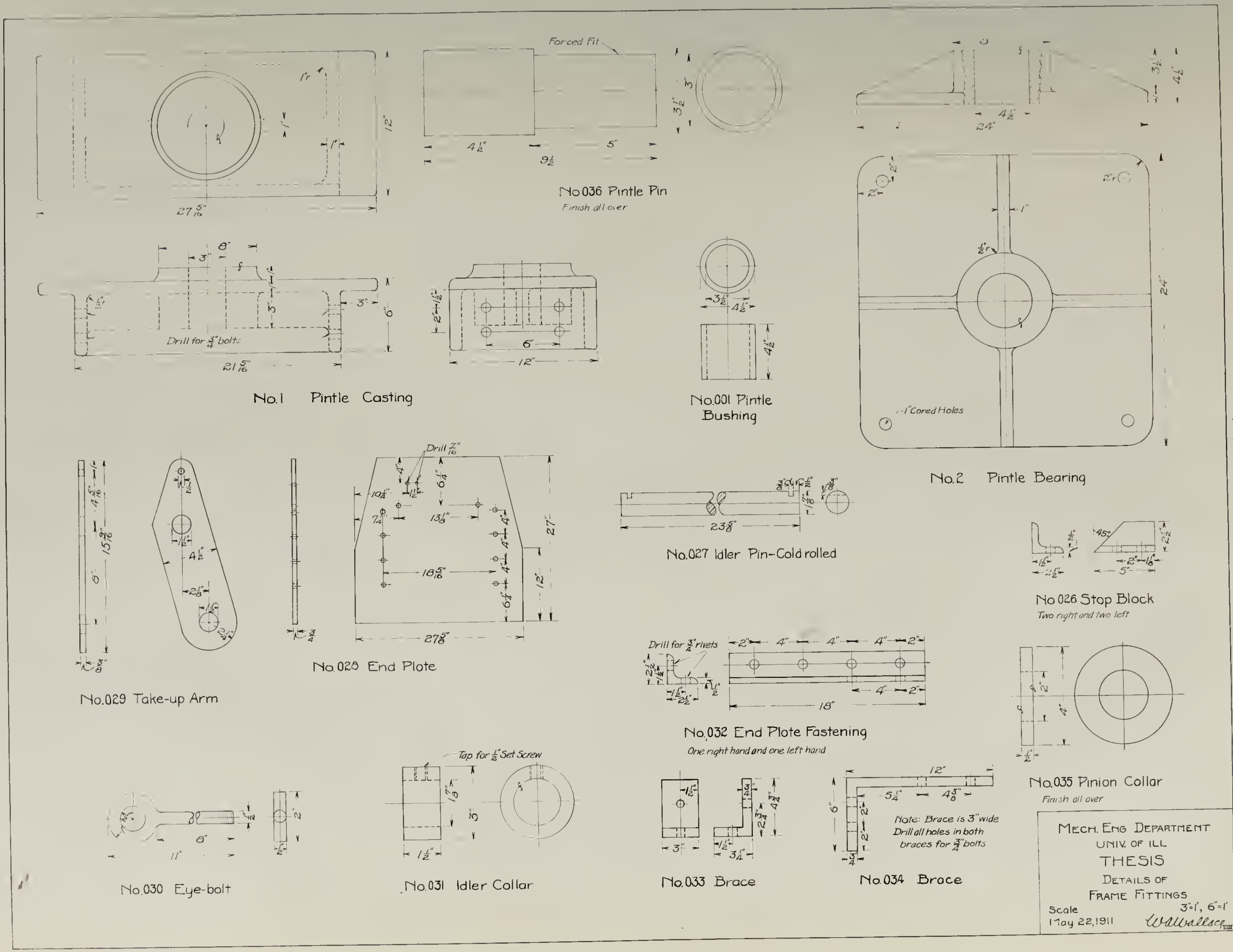

Mech Eng Department
Univ of Ill.
THESIS
Details of
Block
Scale 3"=1'
May 22, 1911
No 11 Sheave
No 015 Side Plate
No 016 Detail of Pin
Drill 13/32
No.012 Pin Lock
No.013 Hook Trunion
Assembly
6 Threads per inch
No.014 Hook


Mech. Eng Department
Univ. of Ill
Thesis
Details of Trolley
Scale 3" = 1' 6" = 1'
May 22, 1911.
No.9 Trolley Axle Bearing
Tapped Holes
No.10 Oil Pocket
Drill 17/32"
No.020 Oil Pocket Cover
No.017 Trolley Axle
Press Fit for Wheel
No.018 Sheave Pin
No.8 Trolley Wheel
Assembly
No.019 Side Plates
For Center Plates only
Note Middle plates to be cut off square as shown by dotted lines
Tap 1/2"

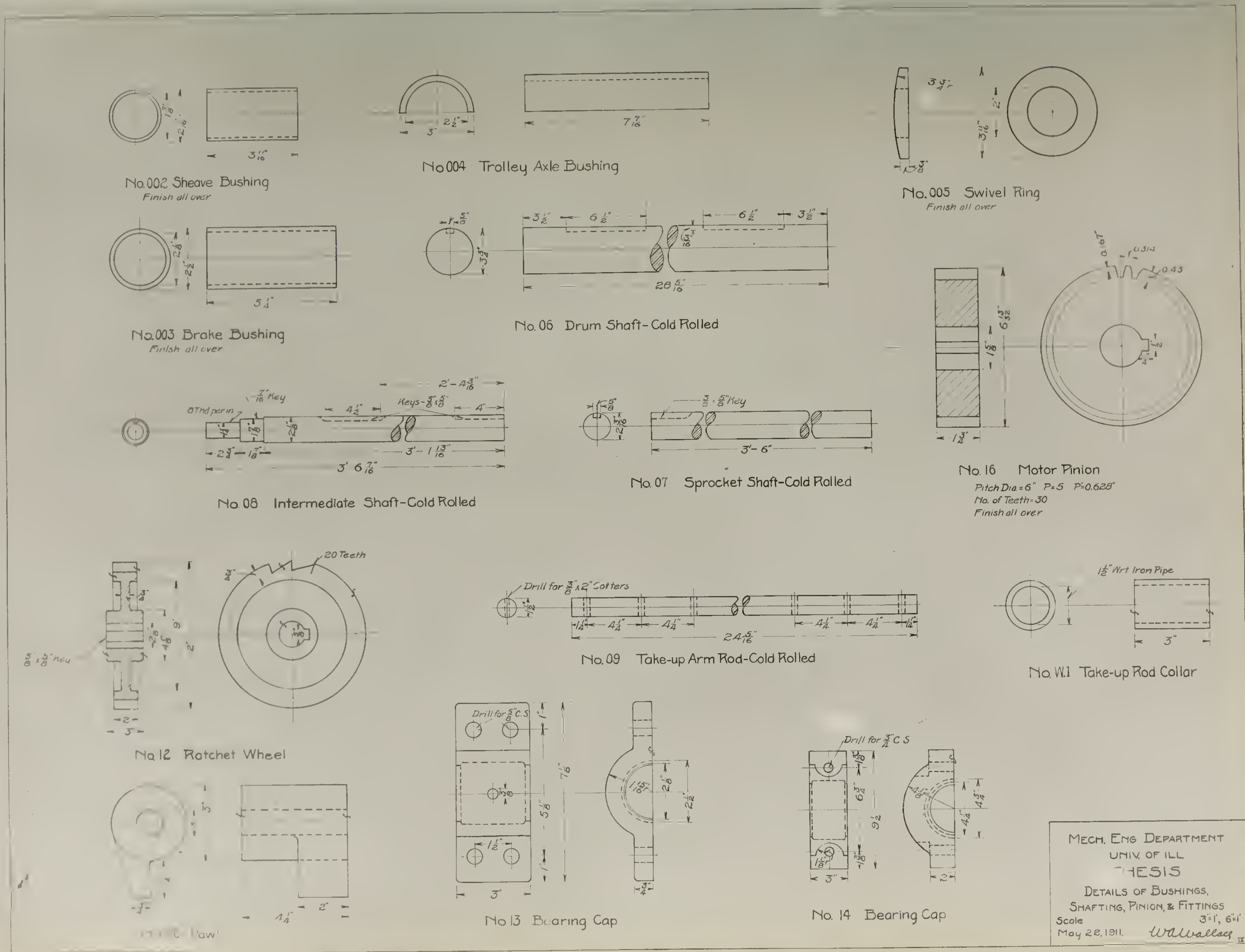


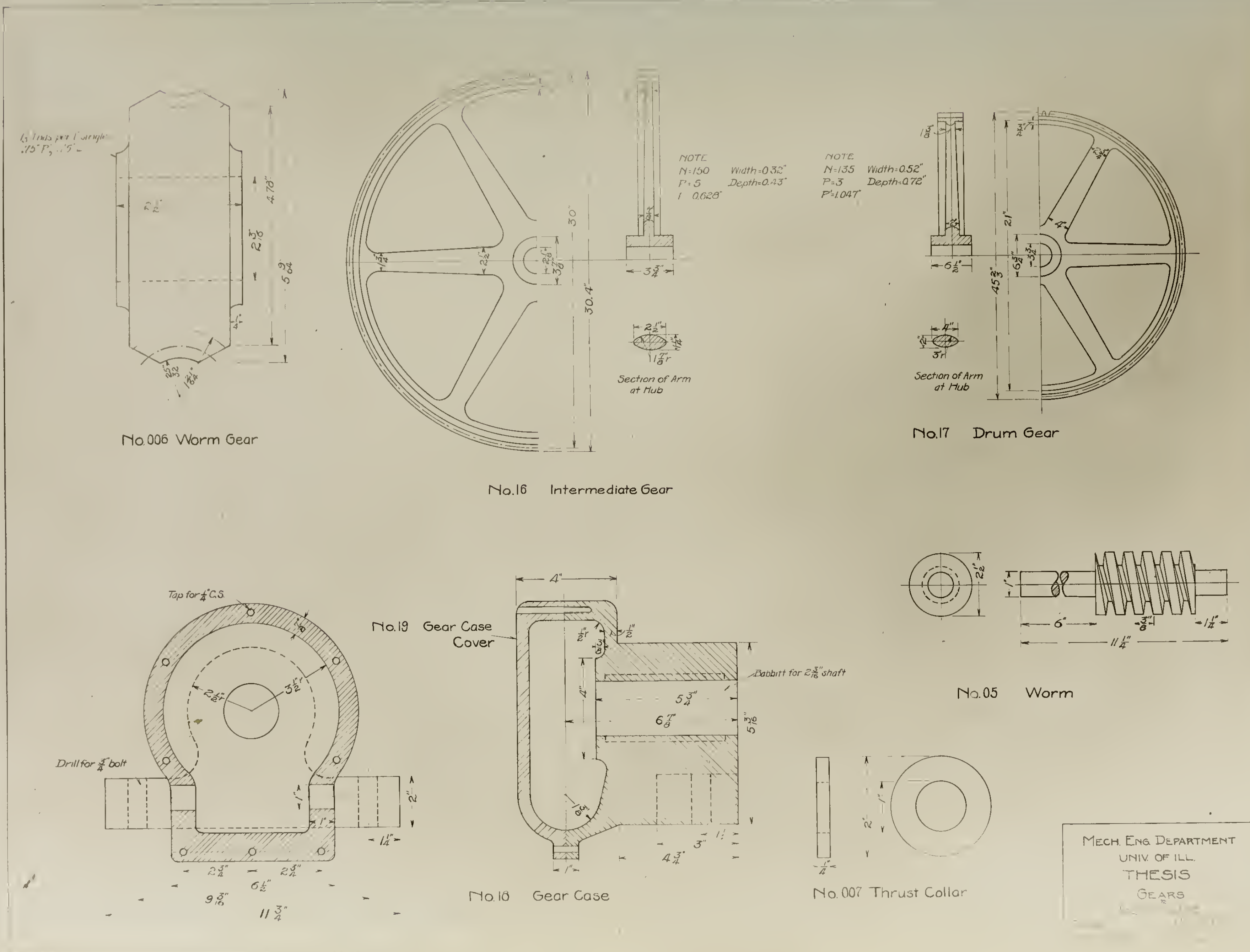
Assembly
No.3 Side Frame
No.022 Tie Bar
Drill for 3/4" bolts
No.023 Motor Brace
Drill for 3/4" bolts
No.024 Motor Brace
Drill for 3/4" C.S.
No.025 Shaft Collar
Tap for 1/2" S. Screw
NoW2 Chain Anchor
Drill for 1/2" Cap Screw
File to fit chain
Full Size Section of Score
22" Pitch Dia.
Tap to fit chain anchor
Drill 7/8"
No.4 Drum
MECH ENG DEPARTMENT
UNIV. OF ILL.
THESIS
DETAILS OF
WINCH
Scales 6"=1', 1 1/2"=1', 3"=1'
May 22, 1911


Assembly
No 5 Brake Casing
Pitch Diam 5"
Diametral Pitch 3
No of Teeth 15
Depth of Tooth .719"
Width of Tooth .523"
Slope 1 in 9
No 6 Hand Wheel
Drill & Tap for 1¼"-8T
No.7 Disc Flange
No.021 Steel Disc
No.F1 Fibre Disc
Mech Eng. Department
Univ. of Ill
THESIS
Details of
Dispatch Brake
Scale 6"
May 22, 1911



No.1 Pintle Casting
Drill for 3/4" bolts
No.036 Pintle Pin
Finish all over
Forced Fit
No.001 Pintle Bushing
No.2 Pintle Bearing
1" Cored Holes
No.029 Take-up Arm
No.028 End Plate
Drill 7/16"
No.027 Idler Pin-Cold rolled
No.026 Stop Block
Two right and two left
No.032 End Plate Fastening
One right hand and one left hand
Drill for 3/4" rivets
No.035 Pinion Collar
Finish all over
No.030 Eye-bolt
No.031 Idler Collar
Tap for 1/2" Set Screw
No.033 Brace
No.034 Brace
Note: Brace is 3" wide
Drill all holes in both
braces for 3/4" bolts
Mech. Eng Department
Univ. of Ill
THESIS
Details of
Frame Fittings
Scale 3"=1', 6"=1'
May 22, 1911
W.A. Wallace


No.002 Sheave Bushing
Finish all over
No 004 Trolley Axle Bushing
No.005 Swivel Ring
Finish all over
No.003 Brake Bushing
Finish all over
No. 06 Drum Shaft-Cold Rolled
No. 16 Motor Pinion
Pitch Dia = 6" P=5 P'=0.628"
No. of Teeth=30
Finish all over
No. 08 Intermediate Shaft-Cold Rolled
No. 07 Sprocket Shaft-Cold Rolled
20 Teeth
Drill for 3/8" x 2" Cotters
1 1/2" Wrt Iron Pipe
No. 09 Take-up Arm Rod-Cold Rolled
No. W.1 Take-up Rod Collar
No 12 Ratchet Wheel
No 13 Bearing Cap
No. 14 Bearing Cap
MECH. ENG DEPARTMENT
UNIV. OF ILL.
THESIS
DETAILS OF BUSHINGS, SHAFTING, PINION, & FITTINGS
Scale 3"=1', 6"=1'
May 22, 1911.
W.A. Wallace


No. 006 Worm Gear
No. 16 Intermediate Gear
NOTE
N=150 Width=0.32"
P=5 Depth=0.43"
1 0.628"
Section of Arm at Hub
NOTE
N=135 Width=0.52"
P=3 Depth=0.72"
P'=1.047"
Section of Arm at Hub
No. 17 Drum Gear
Tap for ¼" C.S.
Drill for ¾" bolt
No. 19 Gear Case Cover
Babbitt for 2 3/16" shaft
No. 18 Gear Case
No. 05 Worm
No. 007 Thrust Collar
MECH. ENG. DEPARTMENT
UNIV. OF ILL.
THESIS
GEARS